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TAPE RECORDER BELT STUDY

Final Report
On

Belt-to-Pulley Creep, Coefficient of Friction, and Stress Relaxation of Seamless Polyester Belts

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SUMMARY

The coefficient of friction, belt to pulley creep, and stress relaxation characteristics of polyester film belts, as typically employed in magnetic tape transmission systems, were determined relative to various physical and environmental factors. A belt dynamometer was developed to measure the torque-speed characteristics of polyester film belt drive systems with a variety of pulley materials and surface finishes. The coefficient of friction was deduced from a characteristic point on the torque-speed curve for each of the material-finish combinations tested. It was found that the coefficient of friction is substantially lower than had been previously reported. The coefficient of friction was found to be independent of the initial surface finish since all the finishes tested were quickly reduced to an equilibrium value of surface roughness. The long term behavior of these drive belt systems was checked by periodically testing a given belt installation over a period of two weeks. The response was quite erratic but there is a strong indication that the capacity of a system drops, during this period, to about 65% of its initial capacity at room temperature but operation at 150°F indicated a very substantial increase in capacity.

INTRODUCTION

Seamless belts which have been fabricated from polyester film are widely employed in instrumentation equipment. This equipment in turn is frequently limited in size and weight (for space applications) with very high reliability requirements. These contradictory aims create designs which are close to the capability of the belt drive systems. The only design information which has been available, up to this time, has been in the report by Licht and White (1). This information is quoted in the report, only for turned metal shafts. The influence of pulley material, and finish upon the coefficient of friction and the change in drive capacity of installed belt with time as a function of temperature and stress are not known.

A study program was undertaken to determine the coefficient of friction with various pulley materials and surface finishes, to determine the effect of operation at a temperature elevated from the installation temperature, and also the effect of time upon the drive capacity. A belt dynamometer was built to measure the drive characteristics of these belts under controlled environmental conditions.

Also, it was felt that the tension in a belt could be deduced from the drive capacity in an installed belt, without disturbing the setup, if the setup were first tested with various installed tensions to establish the relationship between the drive capacity and tension.

To this end all the pulley material, surface finish combinations were tested at room temperature and several installed tensions. Selected test pulleys were then tested at different wrap angles or at different elevated temperatures. After this several test pulleys were retested to determine the variability of the measurements. The last phase of the belt testing was the stress relaxation runs where the belts were set up and the drive capacity measured at increasing intervals for a period of several weeks. Various material on stress relaxation indicates that the loss in stress can be represented as an exponential decay function. In particular, an article by C. L. Carlson (2) indicated that, for steel compression springs, the time constant is on the order of ten days. Some direct measurements of tension in polyester belts under constant strain indicated that the same behavior could be expected. They also indicated that a testing period of two or three weeks should prove adequate to determine the parameters of the exponential decay function.

TEST EQUIPMENT

When flat belts transmit power the driven pulley will run slower than the speed indicated by the geometry of the drive. The difference between the geometrical speed and the actual speed varies in proportion to the torque which is transmitted. This variation from nominal speed is linear with torque at the lower torque values up to a point which is in the range of 60% to 80% of the ultimate torque. The linear portion of the curve is the result of the different elongations in the tighter and slacker sides of the drive belt. See Appendix A for the derivation of the slope of the curve. The remainder of the curve represents the results of incipient slippage over most of the arc of contact and should not be used as a normal drive condition, although it may be useful during transient conditions. The transition between these two portions of the creep-load curve can be used to evaluate the effective coefficient of friction of a belt pulley combination.

The difference in speed of the driven pulley from the speed to be expected from the geometrical ratio is called creep, and is usually expressed in terms of percent of the nominal speed. This usage of the word creep is to be clearly distinguished from the usage which describes the slow permanent change in dimensions of a material under constant load. In this report, the word creep will always refer to the deviation from the nominal speed or motion.

The dynamometer which was designed to make the torque and creep measurements for this study is shown in Figures 1 and 2. The front view, Figure 1, shows the test pulleys and belt installed on the motor and brake shaft. The motor is a 1200/3600 RPM hysteresis synchronous motor with windings rated for operation at 250^{o} F. The entire motor mount can be moved towards and away from the brake to adjust the tension in the belt. The clamp screw can be seen through the hole in front of the mount. On the far end of the machine the brake and torque measuring systems can be seen. The first dynamometer was built with a magnetic particle brake and the three subsequent machines with hysteresis brakes. The overhang of the test pulley on the magnetic particle brake overloaded the bearings so that the bearings had to be replaced every few runs. The construction of the hysteresis brake permitted a reduction in the overhang and the additional shaft support. No bearing difficulties were encountered with hysteresis brakes. The brake was supported in a pair of trunion bearings and carried a weighted arm. The reaction torque of the brake and all running bearing frictions were balanced by the weighted arm which rotated until the load was balanced. A pointer on the arm indicated the torque. The torque is the product of the radius of the center of gravity of the weighted arm and the sine of the angle through which the arm swings. The torque scale was drawn on this basis and each machine was calibrated by means of a balance beam and a set of laboratory weights. With this construction the only frictional force affecting the torque reading is the friction of the trunion oscillating bearings. In operation there is enough jitter present so that the trunion bearing friction torque is averaged out. The load torque is controlled by the voltage supplied to the magnetic brake. A small variable transformer with a step-down transformer and a full wave bridge rectifier was used to vary the field voltage between zero and twenty-four volts. This can be seen in Figure 2. The above describes the means for loading and driving the test belts. The creep was measured by use of a stroboscopic light. A magnetic pickup (See Figure 2) was mounted opposite the tail shaft of the drive motor and provided a trigger pulse, for the strobe light, once per revolution of the motor shaft. Four of the five teeth visible on the sprocket were cut back so that only one pulse would reach the trigger level. This prevents ambiguity in the strobe image. When the strobe light is directed on the driven pulley, the difference in speed between the driving and driven pulleys can be seen as a slow rotation of the driven pulley.

A pattern of four holes can be seen adjacent to each of the test pulleys in Figure 1. Idler pulleys, which are not shown, can be mounted in these holes to change the wrap angles from 180° to 135° and 225°. The tester was also designed to obtain a wrap angle of 90° but the test runs at this wrap angle were erratic and could not be used. This feature was incorporated into the dynamometer so that test runs could be made at several wrap angles as a cross check. The friction torque of these idlers was on the order of the reading accuracy and was not included in the results. All the components were selected for service at 250°F so that the dynamometer could be placed in an oven for elevated temperature testing. For these tests the setup is made at room temperature. The dynamometer and assembled belt are then brought to the elevated temperature as a unit. It was not known whether it would be necessary to allow for the differential expansions of the test fixture and the test belt. This problem was avoided by constructing the test fixture of a material with typical expansion characteristics. The termal expansions of the base of the dynamometer and the belt and the other changes in the physical properties are then the same as would occur in actual service if the housing material is the same as the material of the dynamometer. The base of the dynamometer is aluminum plate, which is frequently employed as a housing or baseplate material in high environment tape recorders. In this way test results at elevated temperatures are directly comparable to actual service conditions with an aluminum structure. Tests were performed at various elevated temperatures when the equipment had stabilized at the test temperature.

The surface finish of the test pulleys was measured before any tests were run and after all the tests had been run. The surface traces which were made before the tests were run were made by Brush Instruments Division of Clevite Corporation. These traces were made with a linear stroke actuator. The stylus lifted and pulled in the opposite direction at the end of each stroke, causing large deflections on the record. The working portion of the trace is marked on the trace. The surface traces which were made after the tests were run were made with rotating work holder and do not show the stylus reversal effects. The surface roughness values obtained from the surface traces are shown in Table I and a pair of actual traces of one pulley is shown in Figure 8. No effort was made to obtain a surface trace on the silicone rubber tire because the stylus would embed in the coating.

TEST PROCEDURE

The test belts used in this program were fabricated from polyester film of 0.002 to 0.005 inch thickness. The width ranged between 1/4 and 5/16 inch and the length was about 22.1 inches. All the belts were fabricated by the drape forming method which was described in the report on fatigue life (3). The belt length is measured with a known load just before it is mounted on the dynamometer. The elongation of the belt under the measuring load is calculated by the expression:

$$e = \frac{PL}{2Etw}$$

e is elongation in inches
P is measuring load pounds
L is measured length inches
E is tensile modulus of elasticity 7.5×10^5 psi
t is belt thickness inches
w is belt width inches

The factor of 2 appears as two cross-sectional areas carry the measuring load. This elongation is subtracted from the measured length to obtain the length of the belt with no load on the belt. This "zero stress length" is the basic length used in all belt stress calculations. Next, the elongation required to provide a given stress in the belt is calculated by the expression:

$$e_t = \frac{s_t}{E} L$$

e_t is elongation in inches for a test
s_t is stress in pounds per square inch for a test

Since each of the tests was run at four stress levels and the length of the belt varies less than 1/2%, these four elongations can be precalculated for the entire run of tests. These elongations are divided by two to determine the required shift of the motor mount.

A starting point for measuring the shift of the motor mount is made by setting the shaft (motor and brake) centerlines 9.429" apart. This makes the path length around the two one inch diameter test pulleys exactly 22.000 inches long. The offset between the motor mount and the end of the dynamometer base is then measured and called the machine index. Twenty-two inches is then subtracted from the zero stress length on the belt and the difference divided by two. This quotient is then subtracted from the machine index to determine the zero stress setting for the belt to be tested. One half the value of e_t is then subtracted from the zero stress setting to obtain the machine setting for a test run. This entire calculation is done in a work form which is on the back of the test data sheet.

The test technician was provided with test data sheets which list the test pulley to be used, the testing speed, the test temperature, and the wrap angle. The test pulleys were fabricated in pairs to each of the material surface finish combinations shown in Figure 3 (Drawing C-12547). The test technician installs the appropriate test pulley pair (the pulleys are installed with two cap screws) and the test belt. The motor mount is then positioned to obtain the desired installed tension in the belt for the test run.

The motor is started and the torque load on the belt adjusted to the first value. This is done by adjusting the voltage to the magnetic brake. With the load set and the strobe light on the driven pulley the speed differential between the driver and the driven pulley can be observed. The time required for some integral number of turns difference is measured with a stop watch. The minimum number of turns counted was two, and as the creep increased, the minimum measured time was two minutes. The test technician then records the torque, the time, and the number of turns on the test data sheet. The torque is increased to the next value and the observation repeated. This process is repeated until the torque limit of the belt is reached. This is seen in either of two ways. The belt may start to slip off the pulley. In this case, a reduction in the torque load will usually catch the belt. If the belt does not slip off the pulley the creep rate becomes so high that it is not possible to count turns. The motor is then stopped and the next installed stress is set up and the data run repeated. Four installed stresses are run in one test on one belt.

The test technician then converts the torque readings into measured torques by use of the calibration curve for the machine, and calculates the change in stress and the creep rate for each observation. The change in stress is calculated by use of the expression:

$$\Delta s = T/dtw$$

Δs is the change in stress from the installed stress in each side of the belt in psi.

T is the torque in lb. in.

d is the pulley diameter in inches (1" dia.)

t is belt thickness inches

w is belt width inches

In this expression the denominator is a constant for a given test. The percentage creep is calculated by:

$c = 100 N/tn$

c is creep in percent of nominal speed

N is number of revolutions counted

t is time for N revolutions in minutes

n is nominal speed in RPM (1200 or 3600 RPM)

The values of Δs and the corresponding values of c are then plotted, the four installed tensions for each test pulley on one sheet. This procedure was written in the form of step by step operations for the test technician. The above procedure provides the data on the torque capacity and belt to pulley creep for each test pulley. It is then possible to deduce the coefficient of friction for each test pulley. Several test pulleys were tested a number of times to determine the variability of the data.

The long term behavior of a belt was determined by an adaptation of the above procedure. The test is set up in exactly the same way as the previous runs and the setting is left undistrubed for the duration of the test. The torque is varied and the creep measured as above on the day of the test, and at the same time of the day the test is set up, on the next day, and then at increasing intervals for a period of three weeks. These "stress relaxation" runs were made under two basic conditions to simulate service conditions. Two types of stress relaxation runs were made. In one of these types the belt was run continuously for the duration of the test. The other type of stress relaxation run left the belt stationary except when the measurements were taken. In both cases there was no torque load on the belts except during the periods when the torque-creep measurements were made. These two types of stress relaxation runs simulated two extremes for long term usage.

The first type simulated long term operation while the second type simulated long term storage followed by operation. This parallel testing was done to establish whether the polyester belts showed a different response under these two situations. It was intended to test the same tension and temperature conditions with both the running and idle belts; however, this proved impractical at elevated temperatures. The motors were supposed to be capable of operation at $250^{o}F$ but when operating at no load these hysteresis synchronous motors dissipated so much energy as resistive heating that the motors could not operate for more than a few hours in a $150^{o}F$ environment. The motor temperature became so high that the housing distorted causing the rotor to drag on the pole faces or the bearings failed by lubricant failure in a few days of operation.

TEST RESULTS

The test data sheet and the plot of test points was turned over to the project manager who drew the curves through the test points. As discussed previously the initial portion of the curve has a uniform slope and at each installed stress starts to deviate at a torque which is a direct function of the installed stress. The coefficient of friction (ratio of maximum frictional force to normal force) to be used for a pulley material-finish combination is deduced from a characteristic point, as the curve starts to deviate from a straight line, and the installed stress. It was originally planned to take this point where the slope had changed by 25%; however, it was found that this point was not well defined. Therefore, the "characteristic point" was selected as the point where the slope had changed 100%. This point is more easily determined and is very close, in value, to the start of the deviation from linearity. These points are tabulated in Table II and IIa, and a sample plot is shown in Figure 4.

It was indicated in the section on test procedure that the installed stresses were precalculated and preset. However, the first sets of data indicated a discrepancy. The coefficient of friction calculated on the basis of the precalculated stress, was found to vary in proportion to the stress. This result is contrary to the definition of a coefficient of friction; however, it might have been caused by a change in the nature of the contact between the belt and the pulley. When the limiting torque stresses were plotted against the elongation of the belt the stress-elongation points were found to lie on a straight line. (Figure 5). This line, extended to the elongation axis, indicated that the precalculated zero stress setting was in error and that the nature of the contact was not the cause of the discrepancy.

This procedure is entirely analogous to that employed in obtaining a stress-strain curve where the curve is projected back to the zero stress level to determine the origin of the strain scale. The intercept obtained in this way was about 0.035 inches which represents half that amount in travel of the motor mount. The original calculation for the machine index setting was repeated independently and did not change. A probable, but not proven, explanation is that this represents backlash and spring in the system which was not properly taken out when the machine setting was determined. When the installed stresses were recalculated from these new elongations consistent values of the coefficient of friction were obtained. The procedure was then:

1. Plot the values of Δs against the nominal elongations. Where runs were repeated the values of Δs were first averaged and the averages plotted.
2. Visually locate and draw a best fit straight line.
3. Determine the elongations for each run.
4. Calculate the tensile stress for each elongation.
5. Add and subtract Δs from the installed stress to get s_1, the tensile stress on the tighter side, and s_2, the tensile stress on the slacker side, respectively. For the runs which were repeated several times use the average value of Δs.
6. Take the ratio s_1 / s_2. This ratio is equal to $\exp \mu \theta$ where μ is the coefficient of friction and θ is the arc of contact in radians.

7. Using a table of natural logarithms determine the value of the exponent. This is the natural log of the ratio.

8. Divide the exponent by the arc of contact in radians to obtain the coefficient of friction.

The results of the above calculation are also shown in Table II. The standard deviation of the ratio s_1 / s_2 was calculated for each material-finish combination for the three repeated runs. The 95% confidence interval of the tension ratio was then calculated by using Student's "t" distribution (the methods used are available in any statistics reference). The coefficient of friction corresponding to each end of the confidence interval was then determined. The means as shown in Table II are shown in Table III with the confidence interval limits which were calculated for them and recommended value for the coefficient is also shown. It is assumed that corresponding confidence intervals can be applied to the test runs which were not repeated. The chrome-plated and hard anodized surfaces show a significant improvement over the stainless steel and plastic pulleys. The silicone rubber tire shows the highest coefficient of the materials tested. It is felt that the coefficient shown in test No. 15 b is a "wild" value. When the data from the report by Licht and White, Figure 2 of that report, was reduced (to compare the two tests) in the same way the coefficient of friction was found to be .133 for 4,000 psi, installed stress on turned metal shafts. Note that the capacity of the drive systems with the silicone rubber tire levels off at an installed tension stress of 3,000 psi. The coefficient is calculated using the three lower installed stresses but not the highest stress. No tests were made to determine whether the onset of this plateau is determined by the total load on the pulley or the belt stress; also no tests were made to determine the effect of tire thickness upon the onset of this plateau.

There is little correspondence between the surface roughness called out on the test pulley drawing (Figure 3) and the values taken from the surface traces. (See Table I). This is the result of measuring the surface in the direction of the machining marks. It is also obvious that the surface roughness initially specified has little influence on the coefficient of friction to be used since the initial differences between test pulley finishes, for a given material, disappears after a relatively short period of operation. Test pulleys -1 and -11 have about 200 hours of operation, while the -2 and -4 pulleys have about 20 hours of operation and the remainder only about 3 or 4 hours. It appears that the relative motion between the belt and the pulley in the tension changing part of the arc of contact brings the surface to an equilibrium surface condition very quickly. This effect was not determined until after the testing period and none of the obvious points which arise were checked. It is not known whether the coefficient of friction changes with the length of operation of the belt (a new belt was used for each run in this test program), nor what the effect of frequent belt changes on the surface was. However, it is felt that the service life of the belt does not influence the coefficient of friction because it was possible to interpose test points in a torque-creep run (before the tension setting was changed to a subsequent value) to check on doubtful measurements, and have the new points fair into the existing points.

Undue reliance had been placed on the precalculated tensions; therefore, in the high temperature runs the belt was not checked at room temperature before being tested at the high temperature. There is thus no direct comparison between the room temperature and high temperature capacity of a given installation; however, if the same value of installed tension is used, at the elevated temperature as would be expected at room temperature the coefficient of friction calculates at substantially the same value and the drive capacity is probably not affected by the change in temperature for the short term exposure.

The stress relaxation runs are treated in a simplified way. The torque limiting values of Δs are determined from the plot of creep versus Δs as above, for each run. The Δs_n value at each succeeding run is then divided by the first value (Δs_1) to obtain the fraction of the initial capacity at the later time. This data is shown in Table IIa. Four room temperature tests are shown in Figure 6 and one test at 150 ^{o}F is shown in Figure 7. Several of these tests were started more than once because of difficulties experienced with the brake or with the motor of the dynamometer. These abortive runs are indicated where the data is usable. It can be seen that the behavior is quite erratic during the test period. It appears that the capacity approaches 65% of the initial capacity for the room temperature tests. The data have not really settled down and it is not established whether the measurement or the belt system is behaving erratically. The test at 150 ^{o}F indicated a significant increase in capacity at the end of the test period and the plot indicates that the change is still occuring.

CONCLUSIONS AND RECOMMENDATIONS

It was found that the coefficient of friction was lower than had been reported previously. Type 303 stainless steel and epoxy and phenolic pulleys indicate a coefficient in the range of 0.075 to 0.080. Hard chrome plate over stainless steel was 0.12 while hard anodized aluminum was 0.13. Pulleys with silicone rubber tires indicated a coefficient of 0.25 but did also exhibit a limitation in capacity. The drive system showed a total capacity at the highest belt tension which was only slightly higher than the capacity at the next lower belt tension tested. It was not determined whether this limiting capacity was a function of the belt stress or the total load on the belt nor was the influence of tire thickness upon this limitation of capacity determined. This material (silicone rubber tire) as a pulley facing is worth more extensive investigation. There does appear to be a slight correlation between surface hardness and coefficient of friction and it might be desirable to investigate other hard facing materials such as one of the carbides or ceramics. In practice the coefficient of friction and the torque capacity are used to establish the installation tension which is required in an application. The method for employing these coefficients is shown in Appendix B.

The search for improvement in coefficient of friction by way of surface finish was found to be of no use. The various surface finishes which were obtained for this test program were found to be substantially identical at the end of the testing. It appears that the surface finish on the pulleys is not an important specification and need only be compatible with the dimensional tolerance required. In any future testing the surface finish need not be variable if the test pulleys are run in for several hours before any measurements are made.

The capacity of the drive system seems to drop to about 65% of its initial value after two and a half weeks at room temperature. At 150^{o}F the capacity starts to rise after one week and appears to be continuing to rise at the end of two and one-half weeks. This increase in torque capacity might be a reflection of increased belt tension. The study on fatigue life (3) of these belts indicated that operation of polyester belts at 200^{o}F did not have a statistically significant effect on the fatigue life. Also the study on the shrinkage of magnetic recording tape (4) indicates that polyester film at these stress levels, undergoes an elongation which is a direct function of the stress, the elevated temperature and the time at temperature. The fatigue life showed that an increase in belt tension should reduce the fatigue life, which did not occur, and the shrinkage study showed that the polyester film should have stretched; therefore, this increase in torque capacity is probably not a result of increased tension. This increase in torque capacity may reflect a useful increase in coefficient of friction at elevated temperatures. The results of the stress relaxation runs are erratic and it would be desirable to have a number of replications for a longer period of time before firm conclusions are drawn. Unfortunately the testing program was scheduled with these tests at the end of the testing period, since a reproducible behavior was anticipated, and there was no time left to expand the number of test runs. Some time was also expended unproductively in the false starts on elevated temperature running tests. The length of the test period should be left open so that the test can be continued until a stabilized value is assured. No systematic difference was observed, between idle and running drive systems, in these stress relaxation runs.

It is not possible, in these tests to separate the effects of tensile stress and coefficient of friction. On the practical level the relative change in capacity, regardless of cause, is of primary interest. It would, on the other hand, be desirable to separate the effect of coefficient of friction and of tensile stress. If the coefficient of friction does, in fact, change with temperature the relative values of the several materials tested may shift. This would be valuable information for tape recorders which were to be operated at elevated temperatures. On the other hand, if the tensile stress is the factor which changes, the effect upon fatigue life may be the important design factor. For these reasons it would be desirable to be able to measure the tensile stress directly, without touching the belt, so that these effects can be separated. If a direct means for measuring belt tension or tensile stress were available it would also be of value in the checkout of assembled recorders.

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TABLE I

Test Pulley Surface Roughness
Microinch Peak-to-Peak Excluding Waviness

Test Pulley		Before Running	After Running
-1	303 S. S. electropolished	6	4
-2	303 S. S. 8 microinch grind	8	4
-3	303 S. S. 32 microinch grind	8	3
-4	303 S. S. 32 microinch liquid hone	16	4
-5	Hard Chrome over electropolished 303 S. S.	8	2
-6	Hard Chrome over 16 microinch grind 303 S. S.	6	3
-7	7076 Alum. .002 Hard Anodized 16 micro-inch machined	24	6
-8	7076 Alum. .002 Hard Anodized 32 micro-inch machined	30	4
-9	Glass Filled Epoxy 16 microinch grind	20	8
-10	Fabric Based Phenolic 16 microinch grind	1 30	15
-11	Silicone Rubber on 303 S. S. Core	Not measured	

TABLE II

Coefficient of Friction and Belt to Pulley Creep Test Results

Test Pulley No. *	Test No.	Δs psi	s_o psi	s_1 / s_2	$\mu\theta$	μ
-1	1	32	319	1.221		
	24	25		1.170		
	25	25		1.170		
	26	25		1.170		
	27	10		1.064		
	37	25		1.170		
	1	255	1,340	1.470		
	24	100		1.161		
	25	80		1.128		
	26	80		1.128		
	27	125		1.208		
	37	180		1.310		
	1	530	2,700	1.489		
	24	190		1.151		
	25	230		1.185		
	26	150		1.119		
	27	225		1.182		
	37	380		1.329		
	1	1,130	6,660	1.409		
	24	515		1.168		
	25	540		1.177		
	26	550		1.180		
	27	720		1.241		
	37	1,160		1.422		
			Avg.	1.2259	.2036	.0647
-2	2	75	319	1.617		
	20	25		1.170		
	21	70		1.560		
	22	60		1.461		
	23	20		1.133		
	2	250	1,340	1.460		
	20	145		1.243		
	21	210		1.371		
	22	155		1.261		
	23	70		1.110		

*All tests were run with 180° of wrap angle and at room temperature unless indicated otherwise in this column.

TABLE II, continued

Test Pulley No. *	Test No.	Δs psi	s_o psi	s_1/s_2	$\mu\theta$	μ
-2	2	360	2,700	1.307		
	20	280		1.231		
	21	370		1.318		
	22	320		1.269		
	23	210		1.159		
	2	860	6,660	1.299		
	20	715		1.240		
	21	840		1.288		
	22	830		1.283		
	23	740		1.250		
			Avg.	1.3014	.263	.0836
-11	11a	160	543	1.835		
	33	160		1.835		
	34	200		2.165		
	35	195		2.120		
	36	170		1.911		
	11a	685	1,560	2.530		
	33	460		1.838		
	34	580		2.180		
	35	440		1.788		
	36	525		2.020		
	11a	1,260	2,920	2.520		
	33	1,090		2.215		
	34	980		2.010		
	35	760		1.688		
	36	930		1.934		
			Avg.	2.039	.7122	.226
	11a	1,425	6,790	1.530		
	33	1,370		1.505		
	34	1,210		1.433		
	35	1,320		1.482		
	36	1,250		1.450		

*All tests were run with 180° of wrap angle and at room temperature unless otherwise indicated in this column.

TABLE II, continued

Test Pulley * No.	Test No.	Δs psi	s_o psi	s_1 / s_2	$\mu\theta$	μ
-3	3	60	319	1.460		
		125	1,340	1.208		
		300	2,700	1.250		
		665	6,660	1.220		
			Avg.	1.2845	.250	.0795
-4	4	80	319	1.670		
		100	1,340	1.160		
		255	2,700	1.212		
		665	6,660	1.220		
			Avg.	1.3155	.274	.0871
-4 (135°)	13	595	6,180	1.212		
		675	7,200	1.205		
		785	8,560	1.184		
		870	12,400	1.150		
			Avg.	1.188	.1723	.0731
-4 (135°)	13 b	335	2,580	1.300		
		430	3,600	1.270		
		660	4,960	1.307		
		1000	8,840	1.254		
			Avg.	1.283	.2492	.1057
-4 (225°)	15 b	50	102	2.925		
		80	1,120	1.153		
		870	2,480	2.080		
		3130	6,350	2.940		
			Avg.	2.274	.8216	.2100
-4 (150°F)	16	210	1,340	1.365	.3111	.0990
-4 (200°F)	17	130	1,340	1.214	.1939	.0617
-4 (200°F)	18	255	1,340	1.469	.3846	.1222

*All tests were run with 180° of wrap angle and at room temperature unless indicated otherwise in this column.

TABLE II, continued

Test Pulley No. *	Test No.	s psi	s_o psi	s_1 / s_2	$\mu\theta$	μ
-5	5	115	319	2.124		
		190	1,340	1.330		
		280	2,700	1.230		
		825	6,660	1.281		
			Avg.	1.4912	.399	.127
-6	6	85	319	1.727		
		180	1,340	1.310		
		300	2,700	1.250		
		845	6,660	1.289		
			Avg.	1.394	.332	.106
-7	7	80	319	1.670		
		195	1,340	1.340		
		460	2,700	1.410		
		1,205	6,660	1.440		
			Avg.	1.465	.381	.1211
-8	8	100	319	1.915		
		250	1,340	1.460		
		440	2,700	1.390		
		1,100	6,660	1.377		
			Avg.	1.5355	.429	.1366
-9	9a	100	985	1.226		
		255	2,000	1.291		
		440	3,360	1.300		
		840	7,250	1.261		
			Avg.	1.2695	.238	.0757
-10	10	140	985	1.211		
		260	2,000	1.300		
		480	3,360	1.332		
		965	7,250	1.310		
			Avg.	1.2882	.253	.0805

(90^o) No Test 12

*All tests were run with 180^o of wrap angle and at room temperature unless indicated otherwise in this column.

TABLE II a

Stress Relaxation Test Results

Test No.	Test Pulley No.	Δ s in psi	$\Delta s_n / \Delta s_1$ *	Time in Days
180^o wrap, continuous running at room temperature 6, 660 psi installed tension.				
29a	-1	1,025	1.00	1/4
		1,250	1.22	1-1/4
		1,075	1.05	3-1/4
29b	-1	1,285	1.00	1/4
		1,370	1.07	6-1/4
		1,310	1.02	9-1/4
		1,280	.995	12-1/4
		870	.687	16-1/4
180^o wrap, idle at 150^oF, 1, 340 psi installed stress				
30	-1	545	1.00	1/4
		630	1.16	1-1/4
30a	-1	420	1.00	1/4
		365	0.869	2-1/4
		425	1.01	5-1/4
		370	0.880	7-1/4
		480	1.14	9-1/4
		590	1.40	13-1/4
		800	1.90	16-1/4
180^o wrap, idle at room temperature, 1, 340 psi installed tension.				
31	-1	220	1.00	1/4
		215	0.976	1-1/4
		110	0.500	3-1/4
		110	0.500	4-1/4
		90	0.409	7-1/4
		150	0.681	11-1/4
		140	0.635	16-1/4

* Δs_n is the value of Δs obtained on the n'th measurement

TABLE IIa, continued

Stress Relaxation Test Results

Test No.	Test Pulley No.	Δs in psi	$\Delta s_n / \Delta s_1$ *	Time in Days
180^{o} wrap, running at 150^{o} F, 1,340 psi installed tension				
32	-1	No test, motor troubles		
180^{o} wrap, continuous running at room temperature, 6790 psi installed tension				
33	-11	1,200	1.00	1/4
		870	0.725	1-1/4
		900	0.750	3-1/4
		1,040	0.867	7-1/4
		790	0.659	10-1/4
180^{o} wrap, idle at room temperature, 6.790 psi installed stress				
34	-11	1,475	1.00	1/4
		1,750	1.188	3-1/4
		1,420	0.964	5-1/4
		1,330	0.902	7-1/4
		1,350	0.915	11-1/4
		1,100	0.746	14-1/4

* Δs_n is the value of Δs obtained on the n'th measurement

TABLE III.
COEFFICIENT OF FRICTION

Test Pulley	Mean	*95% Confidence Limits Upper	Lower	** Recommended Value of
-1	.0647	.0772	.0518	
-2	.0836	.0989	.0680	.075
-3	.0795	.0932	.0651	
-4	.0871	.1005	.0735	
-5	.1270	.139	.115	.12
-6	.1058	.118	.0925	
-7	.1211	.133	.109	.13
-8	.1366	.147	.125	
-9	.0757	.0897	.0616	.075
-10	.0805	.0944	.0665	
-11	.226	.253	.198)	.25

* See Page No. 15

** Based on average of lowest upper limit and highest lower limit

APPENDIX A (Ref. from Page 4)

Derivation of the Creep-Transmitted Torque Relationship

When an elastic belt operates in conjunction with a pulley the two have the same surface speed at the start of the arc of contact. Near the end of the arc of contact the belt elongation accommodates itself to the tension existing on the second side. If a pulley is the power source (driver) the first side will be at a higher tension and the second side at a lower tension. The surface speed of a second pulley (driven) would then be equal to the speed of the slacker side of the belt. Thus

$$V_1 - V_2 = \epsilon_1 - \epsilon_2$$
$$= (\epsilon O + \Delta\epsilon) - (\epsilon O - \Delta\epsilon)$$
$$= 2\ \Delta\epsilon$$

ϵ_1 is unit elongation of tight side

ϵ_2 is unit elongation of slacker side

ϵO is unit elongation at installed tension

$\Delta\epsilon$ is the change in unit elongation

But stress and unit strain are related by:

$$\epsilon = s/E$$

s is stress in psi
E is tensile modulus in psi

Therefore:

$$V_1 - V_2 = 2\ \Delta s/E = \omega_1 - \omega_2$$

$\omega 1$ nominal driven pulley speed

$\omega 2$ actual driven pulley speed

When, for polyester film:

$$\Delta s = 1,000 \text{ psi}$$

and

$$E = 7.5 \times 10^5 \text{ psi}$$

Δs is change in stress in psi

$$\omega 1 - \omega 2 = 2 \times 1,000/7.5 \times 10^5$$
$$= 0.00267$$
$$= 0.267\%/\ 1,000 \text{ psi } \Delta s$$

APPENDIX B (Ref. from Page 18)

(Ref. from Page 18)

Procedure for Determining Required Installation Tension

The following derivation is a slight modification of the derivation which appears in the report by Licht and White (1). At the limiting torque the ratio of tight side to loose side tension is:

$$\frac{T_1}{T_2} = \exp \mu \phi$$

T_1 = tension tighter side, lb.

T_2 = tension looser side, lb.

μ = coefficient of friction

ϕ = wrap angle or arc of contact, radians

and $$\frac{T_1 + T_2}{2} = T_0$$

T_0 = installed tension, lb.

ΔT = change in tension, lb.

M = transmitted torque, lb.

R = driver radius, in.

D = driver diameter, in.

let $T_1 = T_0 + \Delta T$

and $T_2 = T_0 - \Delta T$

then: $M = (T_1 - T_2)R$

$$= \left(T_0 + \Delta T - (T_0 - \Delta T)\right) R$$

$$= 2\ \Delta T R$$

$$\Delta T = \frac{M}{2R} = \frac{M}{D}$$

and

$$\frac{T_1}{T_2} = \frac{T_0 + \Delta T}{T_0 - \Delta T} = e^{\mu \phi}$$

$$T_0 + \Delta T = (T_0 - \Delta T)\, e^{\mu \phi}$$

$$T_0 \,(e^{\mu \phi} - 1) = \Delta T \,(e^{\mu \phi} + 1)$$

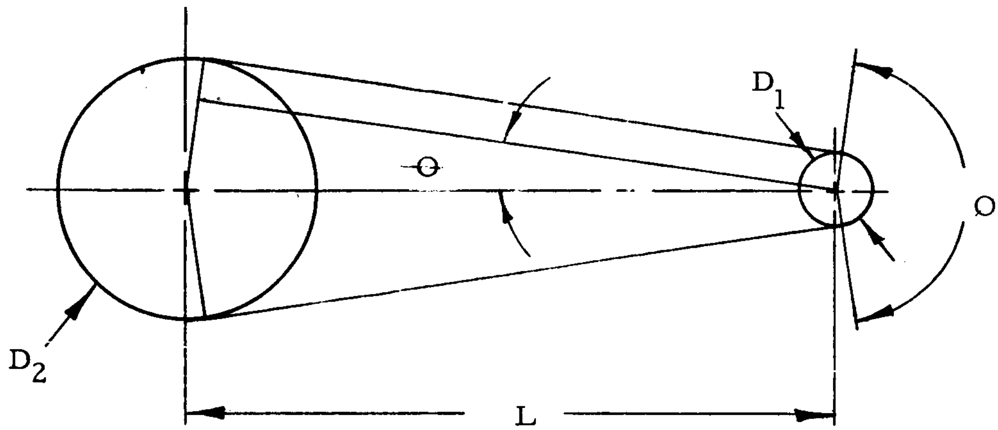
$$T_0 = \Delta T \left(\frac{e^{\mu \theta} + 1}{e^{\mu \theta} - 1} \right)$$

$$= \frac{M}{D} \; \frac{e^{\mu \theta} + 1}{e^{\mu \theta} - 1}$$

The factor $\dfrac{e^{\mu \theta}+1}{e^{\mu \theta}-1}$ is a function of the geometry of the system and the coefficient of friction. This "wrap factor" has been evaluated, in terms of the wrap angle, for several of the coefficients of friction which were established in the program. These values have been plotted on Figure 9. Curves for other values of the coefficient of friction can be drawn in, when required, by simple computation. The procedure is:

1. Read the wrap angle for any given value of the wrap factor on one curve.
2. Multiply this wrap angle by the coefficient of friction for the curve used and divide by the new coefficient of friction.
3. Plot the new wrap angle at the same wrap factor.
4. Repeat for as many values of wrap factor as desired.

An auxiliary angle, called the belt angle, is easily calculated from the geometry of the drive system and has been added to Figure 9 for convenience. The meaning and the derivation of the formula for the belt angle is shown below:



$$\text{S in} = \frac{\frac{D2}{2} - \frac{D1}{2}}{L}$$

$$= \frac{D2 - D1}{2L}$$

FIGURE 9

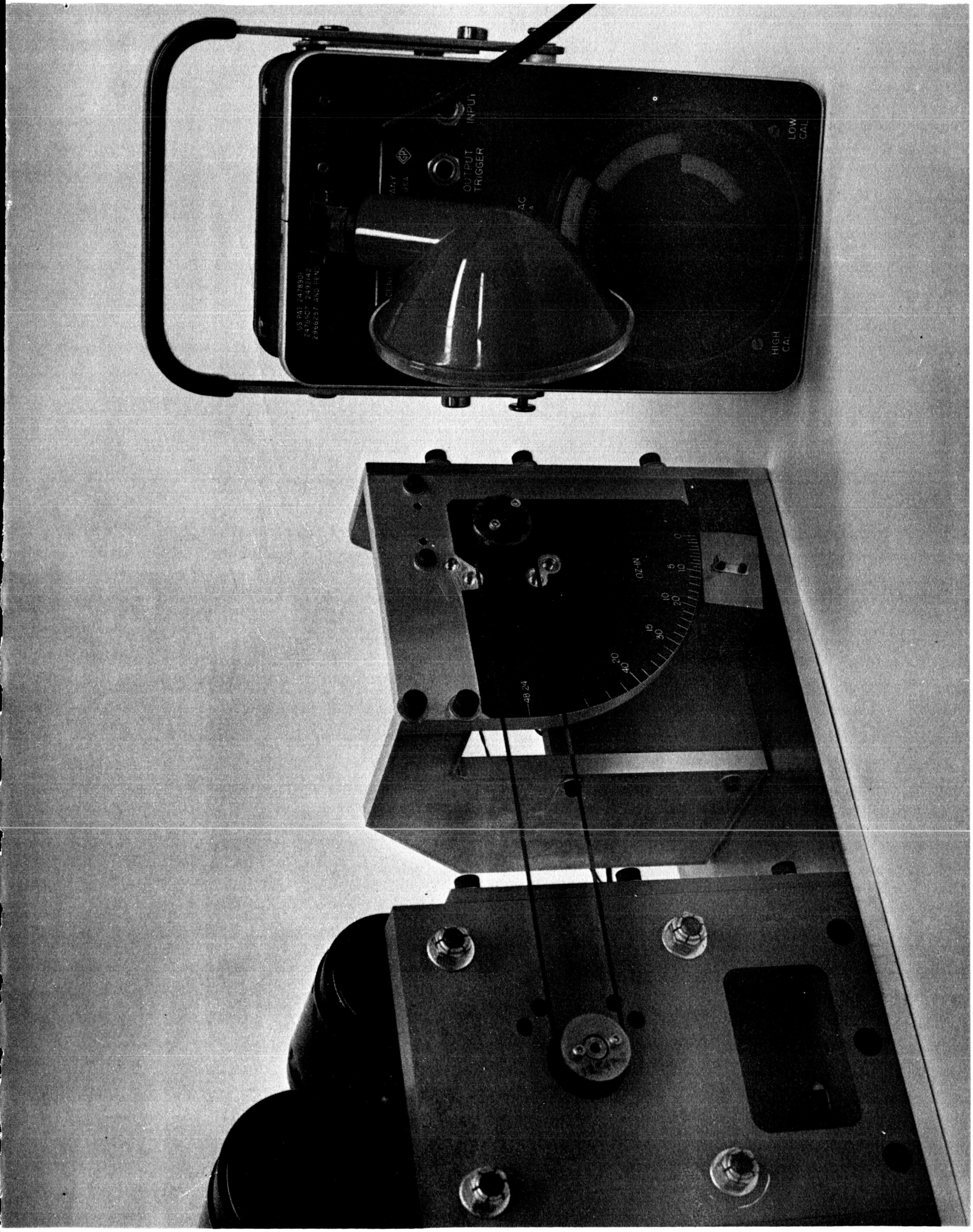


FIGURE 1

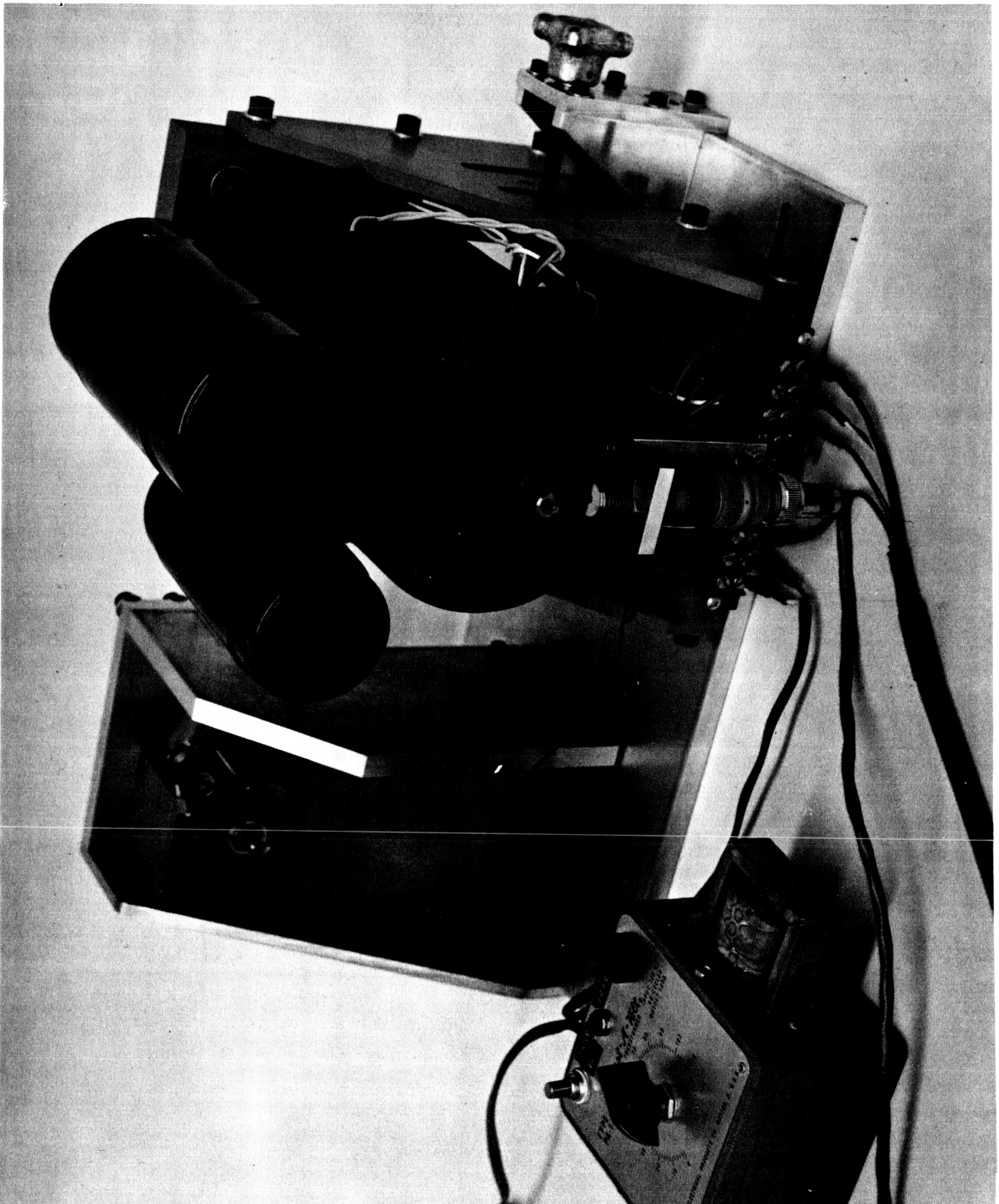
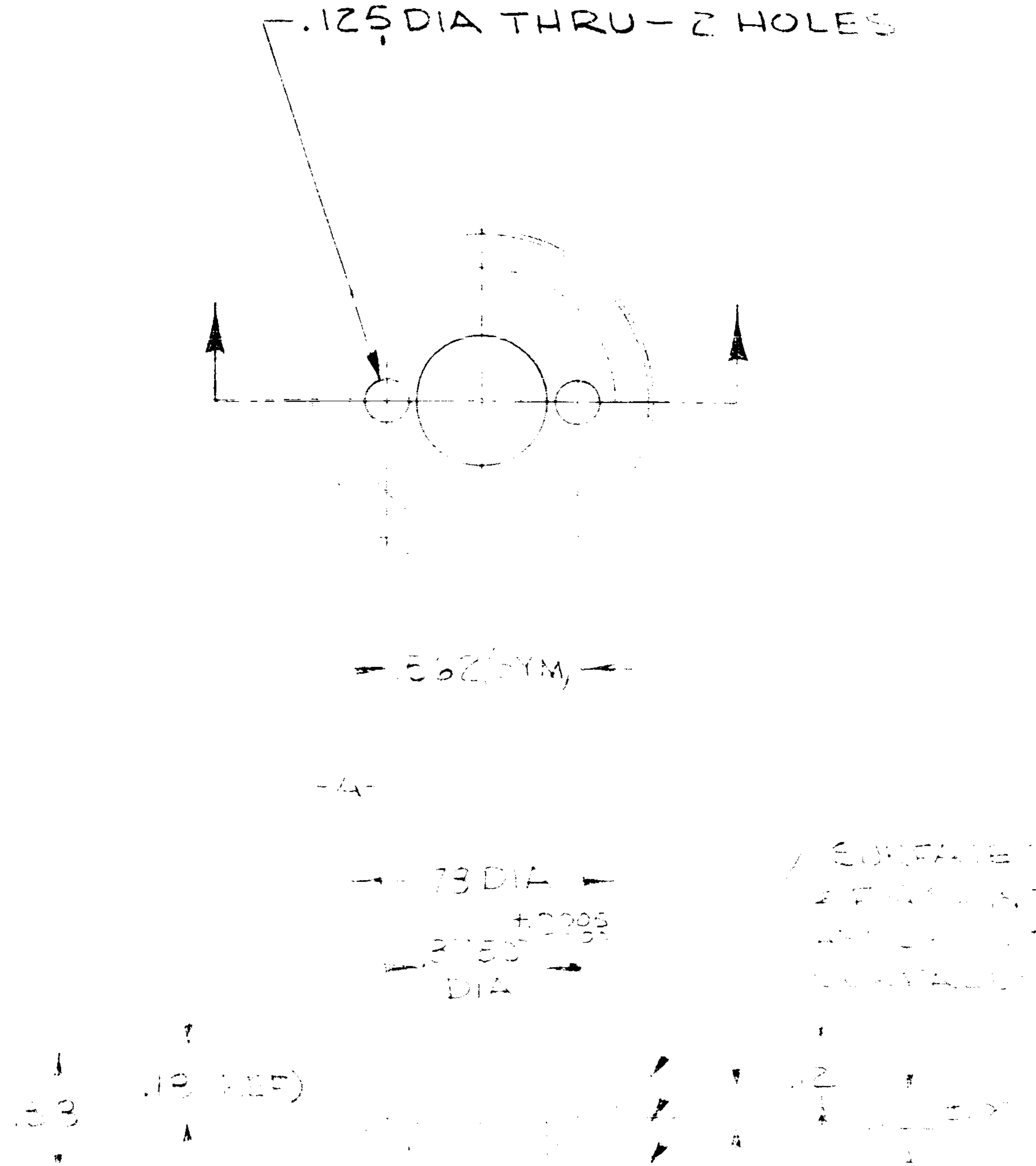


FIGURE 2

1066



TABULATION BLOCK

DASH NO	MATERIAL	SURFACE ROUGHNESS	FINISH
1	CRES TYPE 303	2 μIN ELECTRO POLISH	PASSIVA
2	CRES TYPE 303	8 μIN GRIND	
3	CRES TYPE 303	32 μIN GRIND	
4	CRES TYPE 303	32 μIN LIQUID HONE	
5	CRES TYPE 303	2 μ IN ELECTRO POLISH	HARD CHROM
6	CRES TYPE 303	16 μIN GRIND	ELECTROLIZE
7	AL.ALY. 7076-T6	16 μIN MACH	HARD ANODIZE
8	AL. ALY 7076-T6	16 μIN MACH & LIQUID HONE TO 32 μIN	
9	GLASS FILLED EPOXY NEMA GRADE G-11 FR	16 μIN GRIND	
10	FAB. BASED PHENOLIC NEMA GRADE C		
11	SILICONE RUBBER TIRE PARCO 1122-70	16 μIN GRIND	

RE-ORDER No. 66-95

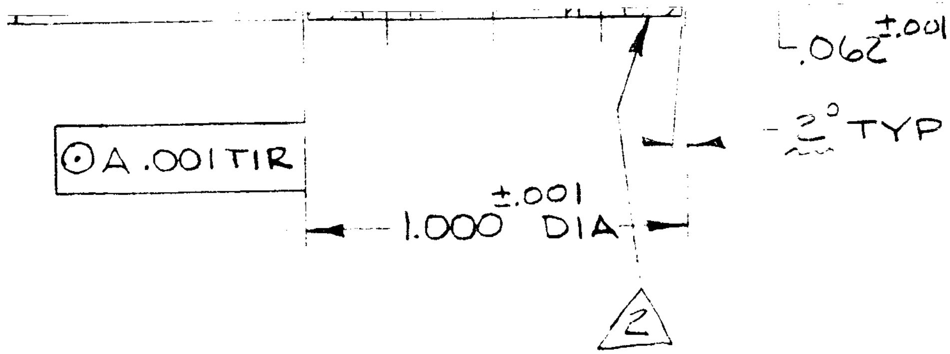
REVISIONS				
CHG	DESCRIPTION	DATE	BY	CHK.

TE

E PLATE
.002THK

.002THK

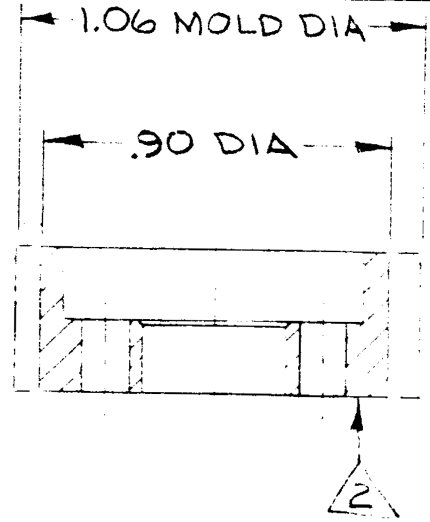
3 of 6



NOTE —

1. DIMS. APPLY AFTER FINISH

2 STEEL STAMP OR ELECTRO-ETCH PART NO ON THIS SURFACE



CORE FOR -11 PULLEY. ALL D
APPLY EXCEPT O.D.

C-12547 | CHG.

MS

TYPE				
NEXT ASS'Y.				

TOLERANCES UNLESS OTHERWISE SPECIFIED

FRAC. ±1/64 1.00 ±.010 1.000 ±.005 ANG	1. CONCENTRICITY .005 T.I.R. 2. BREAK ALL CORNERS .015 MAX. .010 MIN. 3. THREADS CLASS 2 4. FILLET RADII .010 MAX. 5. SURFACES [illegible] MICRO INCH R.M.S. MAX.
MAT'L SEE TAB BLOCK	DATE [illegible] DWN. JONES CHK. APP. DATE
FINISH SEE TAB BLOCK	

DO NOT COPY, DISPLAY OR USE DRAWING WITHOUT AUTHORIZATION. DO NOT SCALE [illegible]

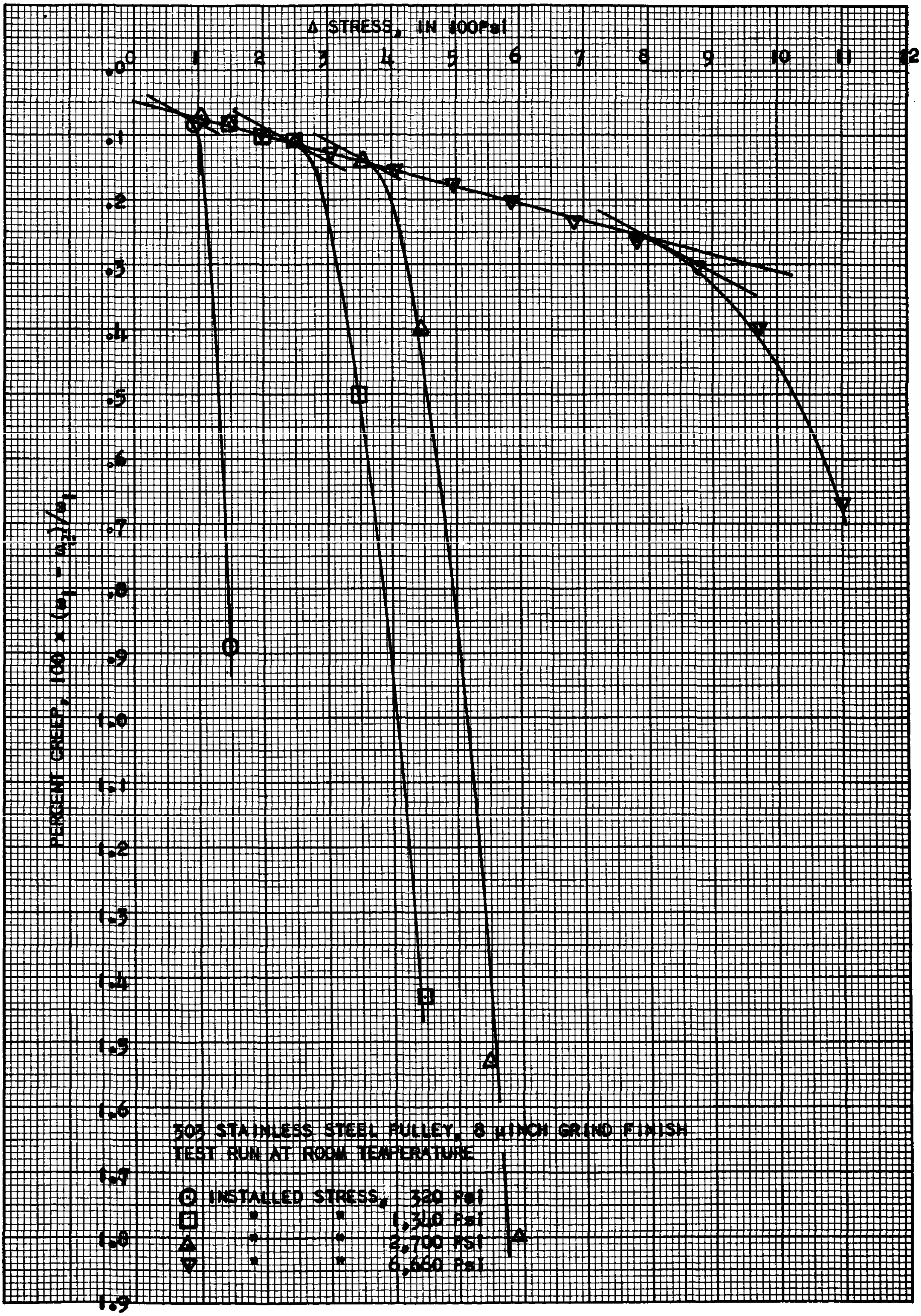
KINELOGIC CORPORATION

PULLEY,
BELT TESTER

2/1 | C-12547

686

F [illegible]



Δ STRESS, IN 100Psi
0
1
2
3
4
5
6
7
8
9
10
11
12
.0
.1
.2
.3
.4
.5
.6
.7
.8
.9
1.0
1.1
1.2
1.3
1.4
1.5
1.6
1.7
1.8
1.9
PERCENT CREEP, 100 x (a1 - a2)/a1
303 STAINLESS STEEL PULLEY, 8 μINCH GRIND FINISH
TEST RUN AT ROOM TEMPERATURE
INSTALLED STRESS, 320 Psi
" " 1,340 Psi
" " 2,700 Psi
" " 6,660 Psi

FIGURE 4

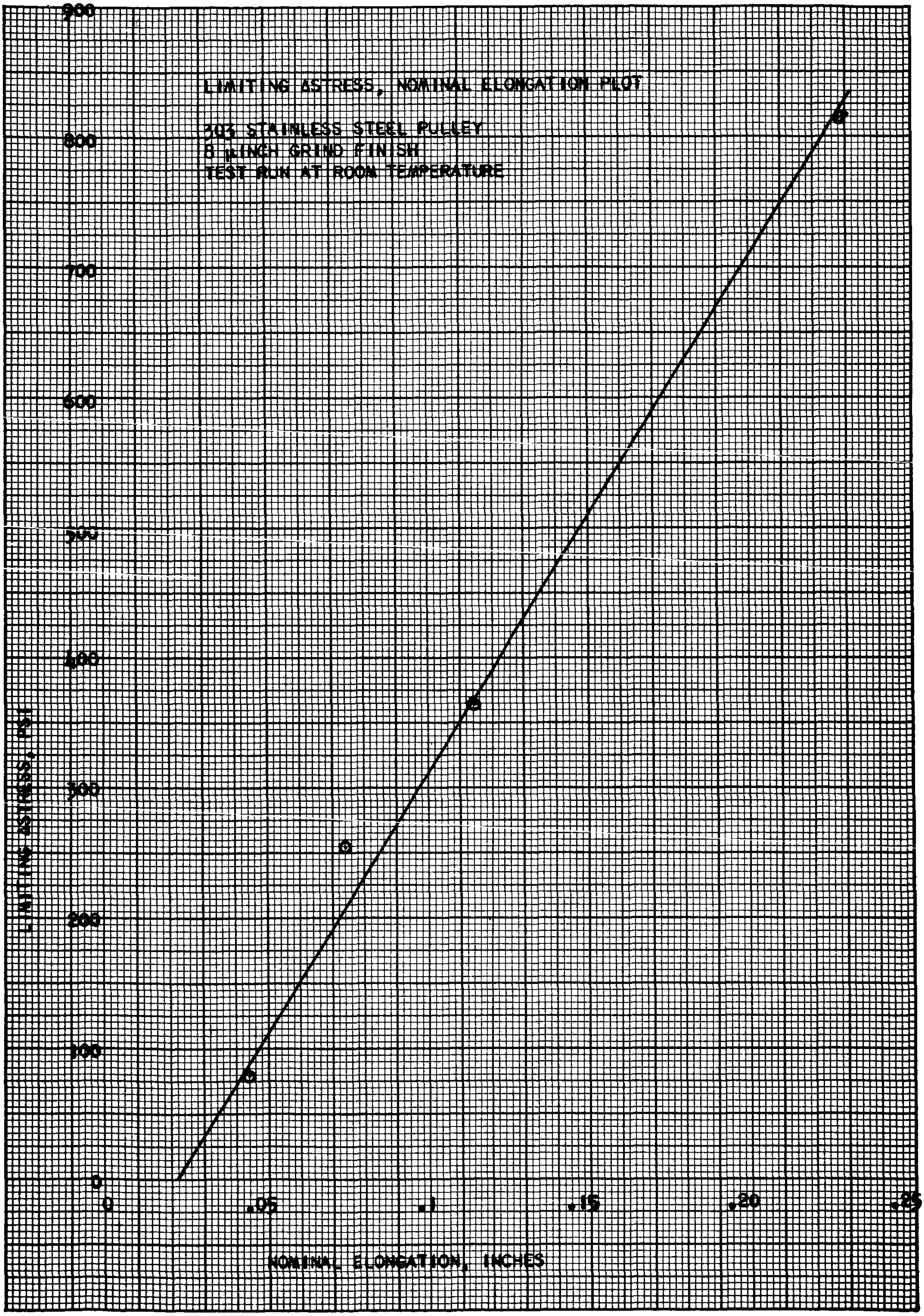
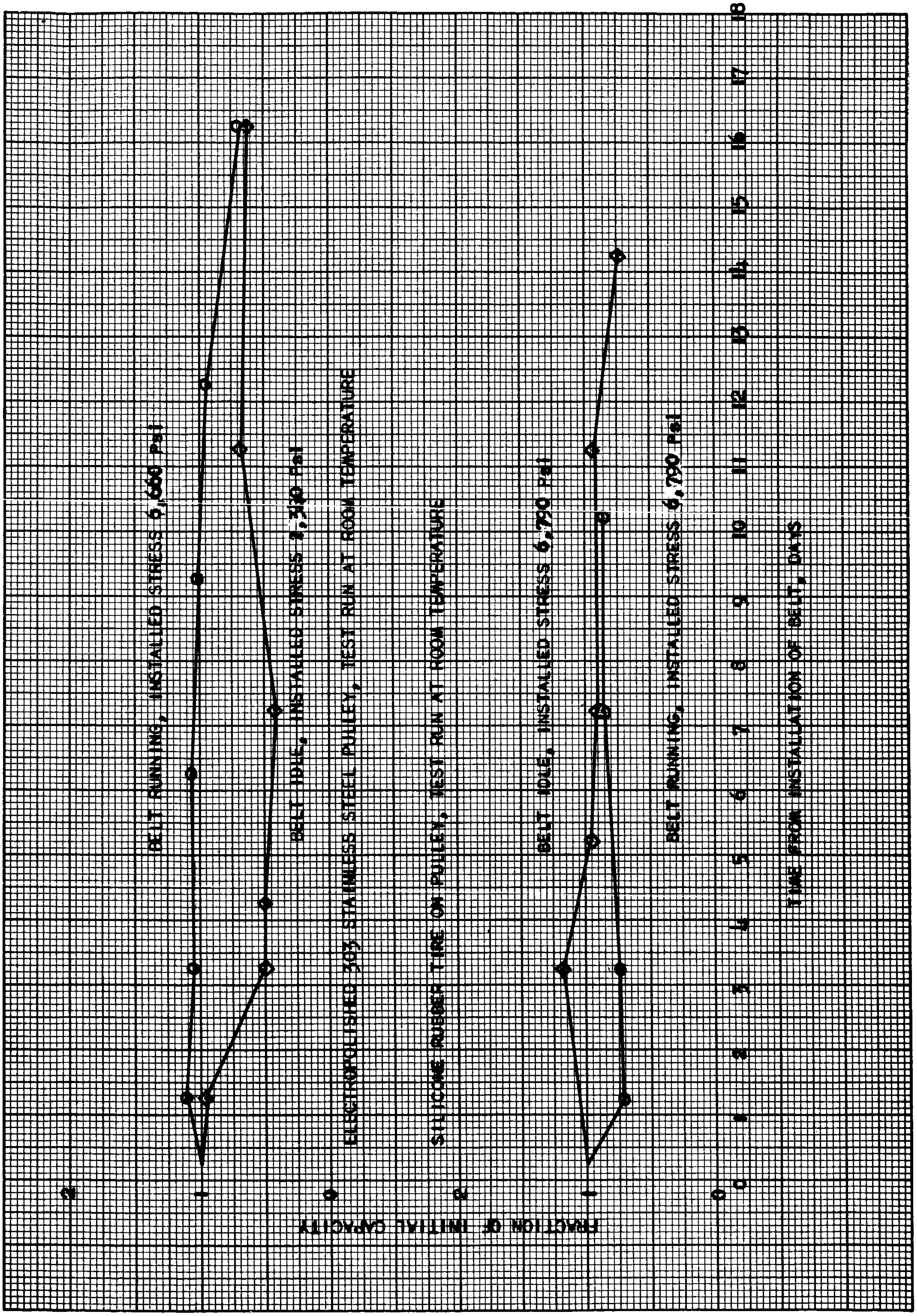

LIMITING ΔSTRESS, NOMINAL ELONGATION PLOT
303 STAINLESS STEEL PULLEY
8 μINCH GRIND FINISH
TEST RUN AT ROOM TEMPERATURE
900
800
700
600
500
400
300
200
100
0
0
.05
.1
.15
.20
.25
LIMITING ΔSTRESS, PSI
NOMINAL ELONGATION, INCHES

FIGURE 5



BELT RUNNING, INSTALLED STRESS 6,660 Psi
BELT IDLE, INSTALLED STRESS
ELECTROPOLISHED 303 STAINLESS STEEL PULLEY, TEST RUN AT ROOM TEMPERATURE
SILICONE RUBBER TIRE ON PULLEY, TEST RUN AT ROOM TEMPERATURE
BELT IDLE, INSTALLED STRESS 6,790 Psi
BELT RUNNING, INSTALLED STRESS 6,790 Psi
TIME FROM INSTALLATION OF BELT, DAYS
FRACTION OF INITIAL CAPACITY
0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18

FIGURE 6

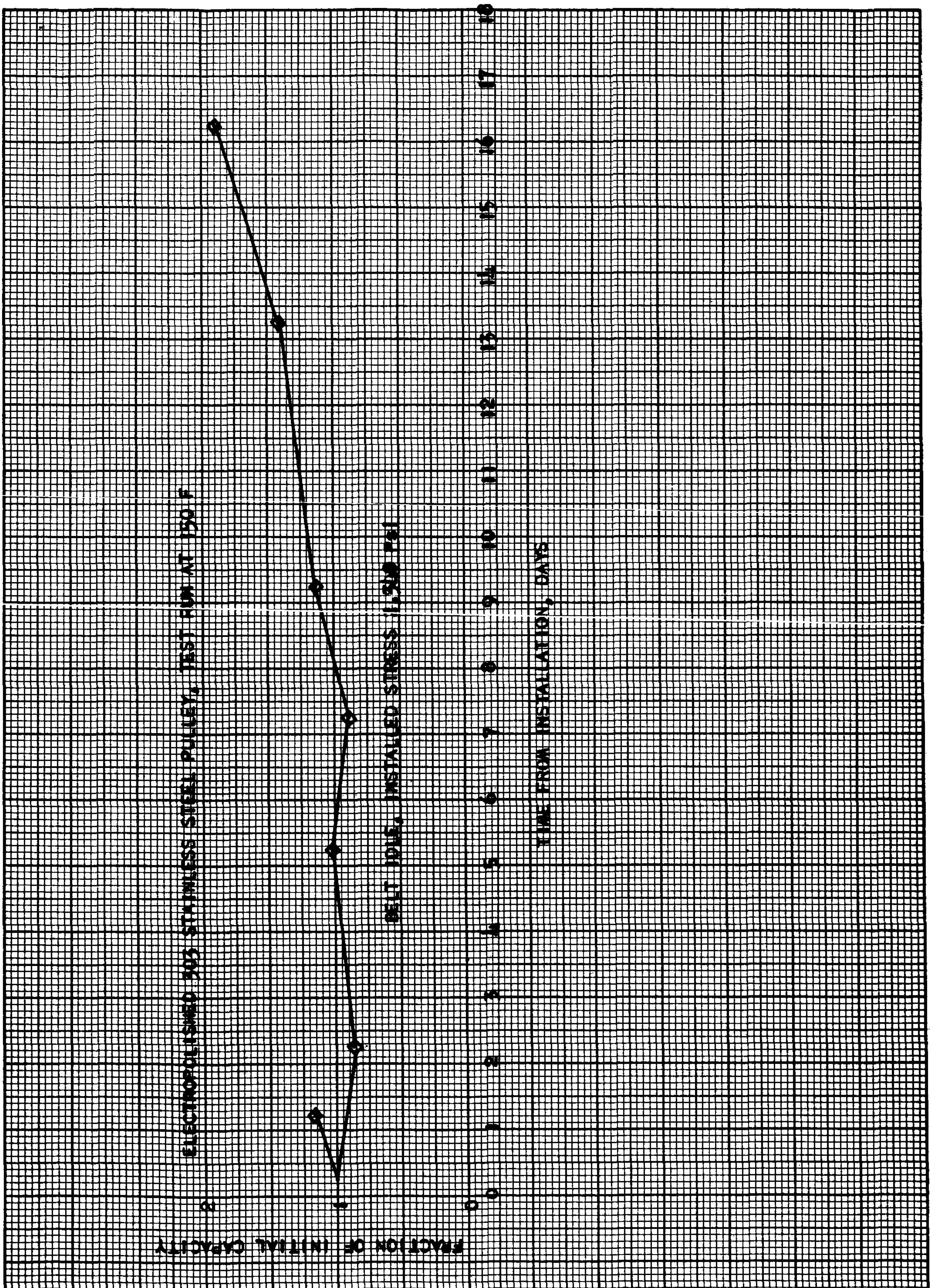
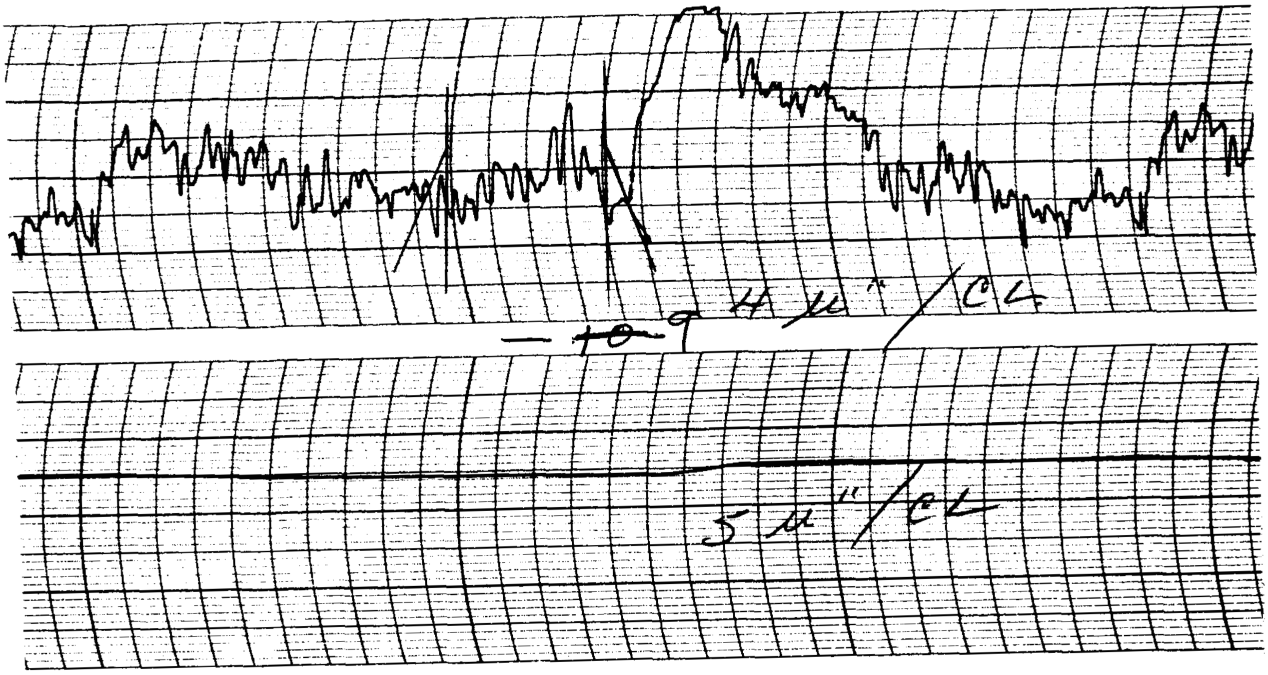
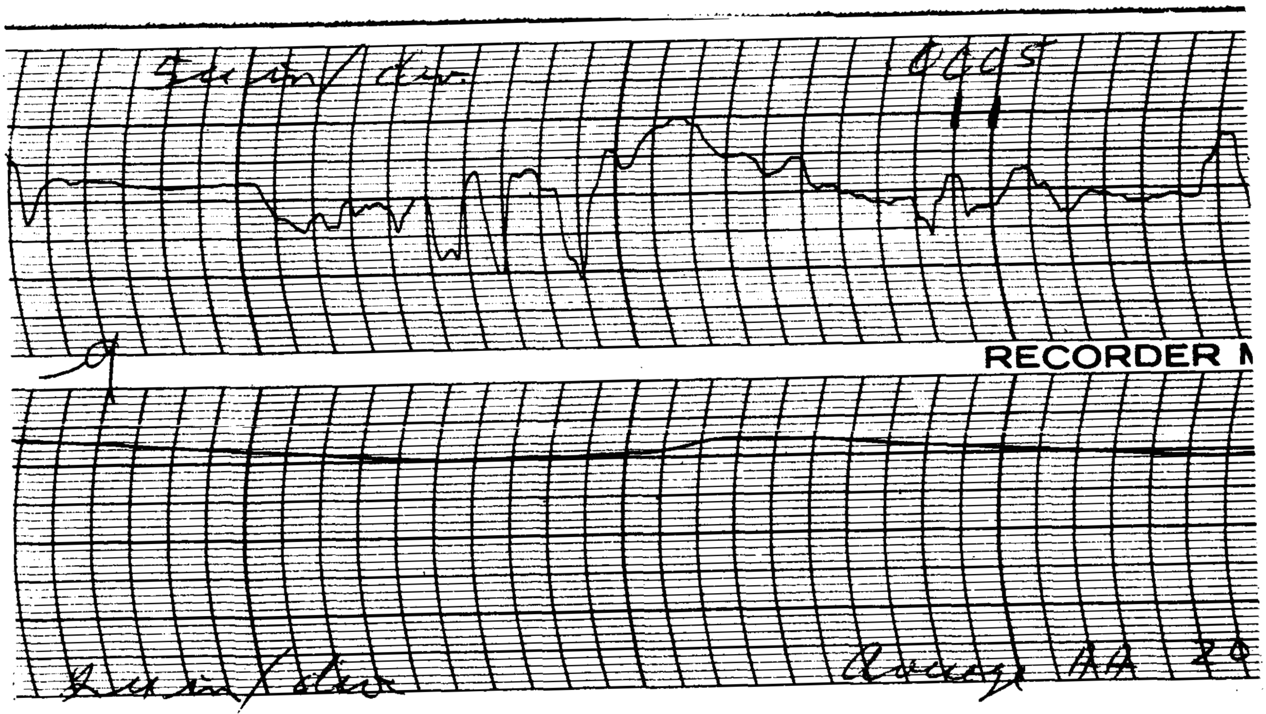

ELECTROPOLISHED 303 STAINLESS STEEL PULLEY, TEST RUN AT 150 F
BELT IDLE, INSTALLED STRESS
TIME FROM INSTALLATION, DAYS
FRACTION OF INITIAL CAPACITY
0
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
17
18

FIGURE 7



BEFORE OPERATION



AFTER OPERATION

-9 PULLEY, GLASS FILLED EPOXY

FIGURE 8

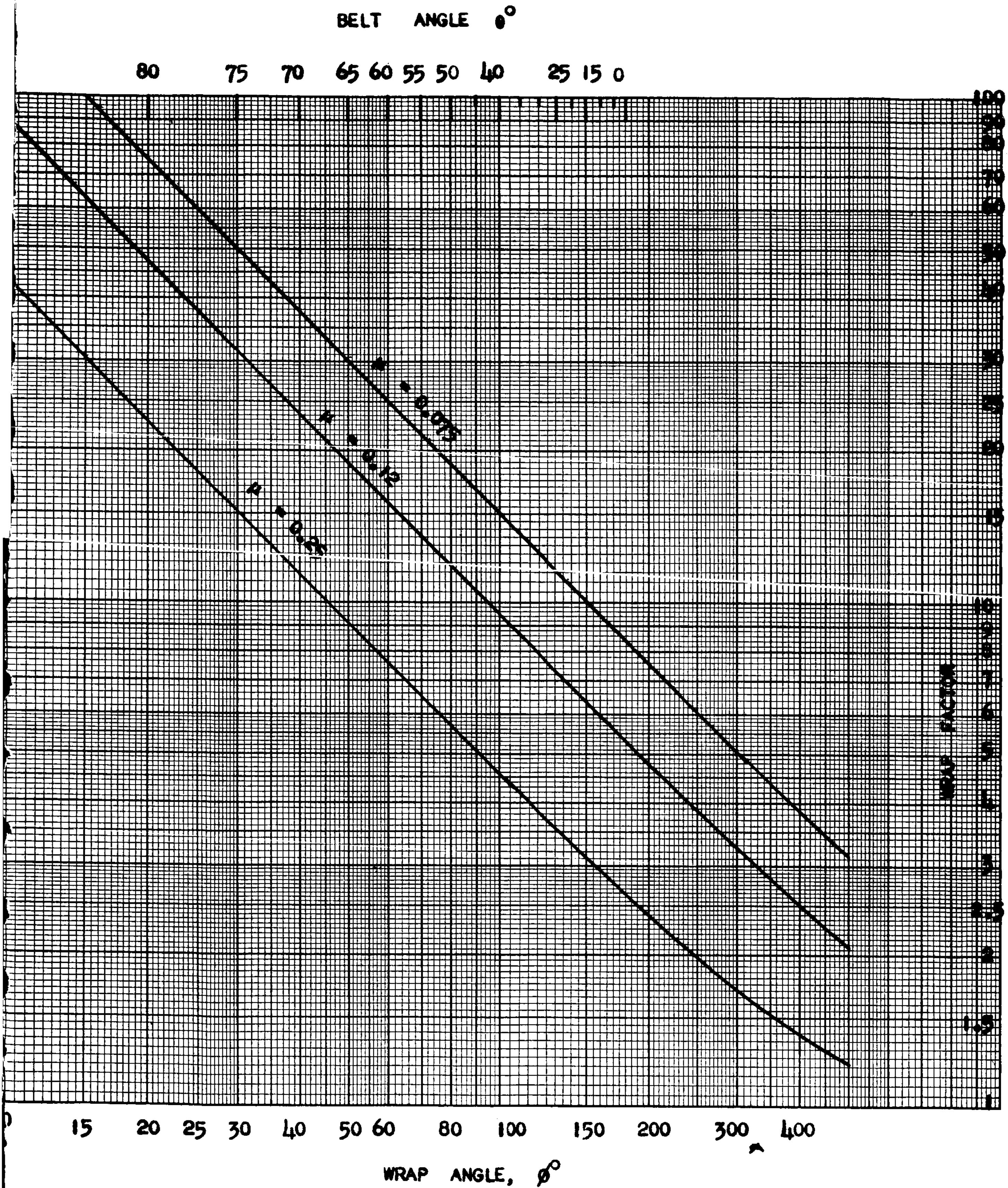

BELT ANGLE θ°
80 75 70 65 60 55 50 40 25 15 0
R = 0.075
R = 0.12
R = 0.25
WRAP FACTOR
15 20 25 30 40 50 60 80 100 150 200 300 400
WRAP ANGLE, φ°

FIGURE 10